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DESIGN AND FINITE ELEMENT ANALYSIS OF HIGH SPEED COMPRESSOR GEARBOX UNIT

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ABSTRACT

The project is concerned with the design and analysis of some of the key elements of a gearbox that is used to drive compressor units. Since the pinions rotate at high speeds the complete gear system is subjected to noise and vibrations. The gear system is composed of one bull gear and two pinions. The helical gears are designed as per AGMA standard procedure. Conventional parallel axes helical gears with Involute profile teeth are insensitive to center distance assembly errors and possess line contact under design assembly condition. However, helical gears are very sensitive to axial misalignments, causing discontinuous transmission errors and edge contacts, resulting in noise and vibrations. Therefore the helical gear are modified to attain a localized point contact and to avoid edge contact. The modified helical gear set contains conventional Involute profile, with profile crowning as well as longitudinal crowningThe geometrical models of helical gear are drawn in CATIA V5. Since, the gear system rotates at high speed; this may cause high imbalance forces and vibrations due to the shaft assembly. So, the shafts are designed as integrated shafts for two pinions and a shrink fit assembly for bull gear and shaft. The non-linear finite element contact analysis of helical gear tooth is done using finite element software, ANSYS. workbench 14.5.7. For economy, different models of gear are checked and finally single tooth contact analysis is done for different positions of gear contact.

KEYWORDS: Helical Gear Design AGMA, Modeling CATIA – V5 Finite Element Analysis ANSYI

INTRODUCTION

Gears are toothed wheels or multi-lobed cams, which transmit power and motion from one shaft to another by means of successive engagement of teeth. Gear drive is a positive drive and maintains constant velocity ratio. It can transfer very high power. As the requirements are broad and are of varying difficulties, gearing is a complex and diversified engineering field. Therefore modifications on the design of gear are suggested depending upon the different applications. The purpose of modification is to improve the performance of the gear. Also gear is machine element which, by means of progressive engagement of projections called(teeth) transmits motion and power between two rotating shafts, gear teeth in general have an involute profile which provides a constant pitch line velocity. The action of such mating gear teeth consists of a combination of rolling and sliding motions, thus producing a positive drive

CLASSIFICATION OF GEAR

The gear are classified on the following basic (1) relation between axes. (2) shape of the solid on which teeth are

cut. (3) curvature of the tooth profilegenerally, the following type of gears are most commonly used in industry for power transmission purposes

Spur Gear: A gear having straight teeth along the axis is called the spur gear, spur gear are used to transmit power between two parallel shafts as shown in figure (1-a) A rack is a straight tooth gear which can be thought of as a segment of spur gear of infinite diameter

Helical Gear: They are also used to transmit power between two parallel shafts and teeth are cut on the cylindrical disc. The tooth faces of these gears have a certain degree of helix angle Angle of opposite hand on pinion and gear as shown in figure (1-b). these gear are smooth in operation and therefore can transmit at a high pitch line velocity.

Bevel Gear: When power is to be transmitted between two intersection shafts, bevel gear are used the angle of intersection of shafts is called (the shaft angle) the gear blank is a frustum of cone on which teeth are generated the teeth are straight but their sides are tapered so that all lines, when extended, meet at a common point called the (apex of the cone) as shown figure (1-c).

Worm and Worm Gears: In this system of gearing, the axis of the power transmitting shafts are neither parallel nor intersecting but the planes containing the axes are generally at right angles to each other. The teeth used are helical the schematic diagram of a worm gear set is shown in the figure (1-d).

In a gear drive, the smaller of the two gears in mesh is called (pinion)and the larger gear is customarily designated as (gear). In most of the applications, the pinion is the driving element whereas the gear is the driven element. [18]

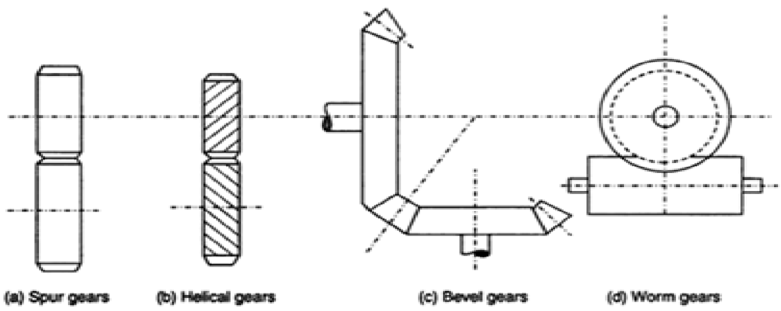

Figure 1

Helical gears normally used for high speed, high power mechanical systems therefore the helical gear used to (1) high speed (2) less noise and vibration (3) used to transmit high power (4)high efficiency 98……..99.5 why?? (in a helical gear drive, the contact begins at the tooth end and as the rotation progresses, the contact point moves along whole tooth width till it reaches the other end this result in a gradual, even tooth action and load distribution unlike spur gear drive, the contact line runs diagonal from one end to the other end of the helical tooth besides, in a helical gear drive, more than one pair of teeth are always in mesh this, and others characteristics, like shorter lever arm, allow the helical drive to have considerably more load carrying capacity.) [4]

HELICAL GEAR

In helical gears, the contact between meshing teeth begins with a point on the leading edge of the tooth and gradually extends along the diagonal line across the tooth. There is a gradual pick up of load by the tooth, resulting in

smooth engagement and quiet operation even at high speeds. Compared to spur gears of the same size, helical gears can transmit more power and are less noisy. In views of this, helical gears are more efficient and are usually desired for high power transmission high pitch line velocity these gears areused in high speed applications helical gears are of two types- parallel gears(single and double) used for parallel shafts and cross- helical gears used for non- parallel shafts[3]

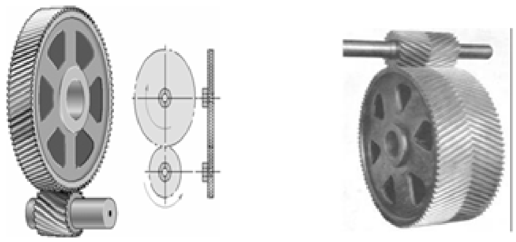

Figure 2

NOMENCLATURE OF THE HELICAL GEAR

Helical gears are also called as prompt gears, prompt gears are used to transmit rotary motion between parallel shafts[8]. These are usually cylindrical in shape and its teeth are have certain degree of helix angle. In a couple of gears, the major one is often called the GEAR and, the minor one is called the PINION

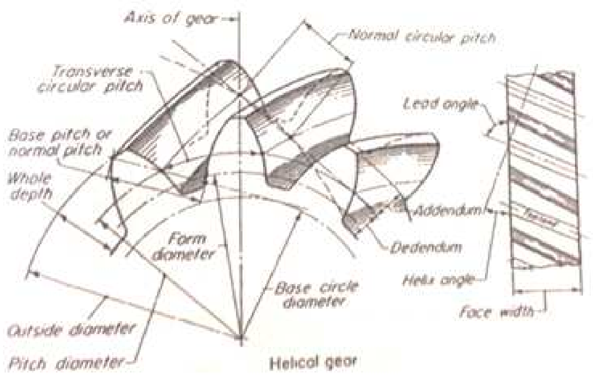

Figure 3

DESIGNING OF GEAR

There are different standards for design of gears, such as German DIN standard, Japanese JS standard and ISO standard. But, the AGMA (American Gear Manufacturer Association) standard is followed for most of the gear design. In this design, the helical gears are designed according to the AGMA standards. On the basic tooth proportions, the AGMA sponsors a large number of standards dealing with gears design, specification materials, ratings and inspection. The most important AGMA standards are

"Gear Handbook" 390.03, "Gear Classification and Inspection Handbook" 2000-A88

The LEWIS Bending Equation

In design of gears, It is required to decide the weaker between pinion and gear. When the same material is used for pinion and gear, the pinion is always weaker than gear [6]

$$\boxed{Y = 0.154 - \frac{0.912}{Z_E}}$$

According to AGMA standard procedure

Beam Strength,

$$S_t = \frac{F_t \times K_a}{K_v} \times \frac{1}{b \times m_n} \times \frac{K_s \times K_m \times K_b}{J}$$

$$S_t = \frac{F_t \times K_a}{K_v} \times \frac{1}{b \times m_n} \times \frac{K_s \times K_m \times K_b}{J} \quad \left[380 \geq \frac{2833.4 \times 1.75}{0.646} \times \frac{1}{55 \times m_n} \times \frac{1 \times 1.2}{0.154 - 7.52 \times 10^{-3} \times m_n} \times \frac{1 \times 1}{1}\right]$$

mn=3.439mm

STRESS CALCULATIONS OF GEARS

Bending Stresses

$$S_t = \frac{(F_t \times K_a \times K_s \times K_m)}{(K_v \times b \times m_n \times J)}$$

= 93.55MPa

Contact Stress

$$S_C = C_P \times \sqrt{\frac{W_t \times C_a \times C_s \times 0.93 \times C_m \times C_f}{C_V \times d \times b \times I}}$$

Sc = 459.128 MPa

Static Tooth Load

$$F_s = \sigma_e \times b \times \pi \times m_n \times Y$$

Wear Tooth Load

$$F_W = \frac{D_1 \times b \times Q \times K}{\cos^3 \psi}$$

Dynamic Effect

$$F_D = F_t + \frac{21 V_1 \times (b \times C \times \cos^2 \psi + F_t) \times \cos \psi}{21 V_1 + \sqrt{b \times C \times \cos^2 \psi + F_t}}$$

MODELING

Gears modeling is very useful and important, as to make real gear transmission simulation. CATIA stand for Computer Aided Three Dimensional Interactive Application. It is a product modeling software can be custom-made via Application Programming Interfaces (API). CATIA V5features are parametric solid/surface-based wrap up which uses NURBS as the interior surface representation and has quite a lot of workbenches that make available KBE support.

The most widely used software in modeling now a days is CATIA.

It can be appreciable to be considered as an User friendly software and capable of presenting the existence on any platform which is related to computer part of programming the commandsThere by accuracy of the software for drafting is almost reaches to preferable real time values. Though there are so many versions in CATIA the V5 version has been used for present work. For helical gear in CATIA, relation and equation modeling is used. Relation is used to express dependencies among the dimension needed for defining the basic parameters on which the model is depends

In this work, module, pressure angle, numbers of teeth of bull gear is taken as input parameters. CATIA V5 uses these parameters,

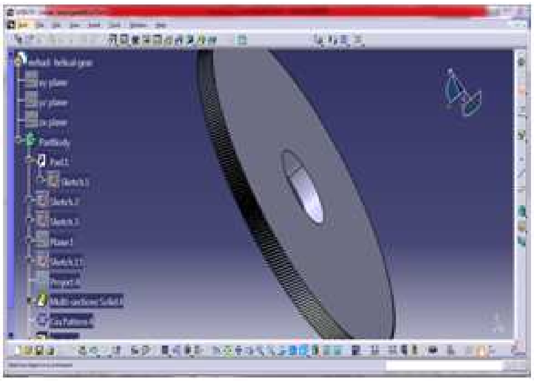

Figure 4: Bull Gear

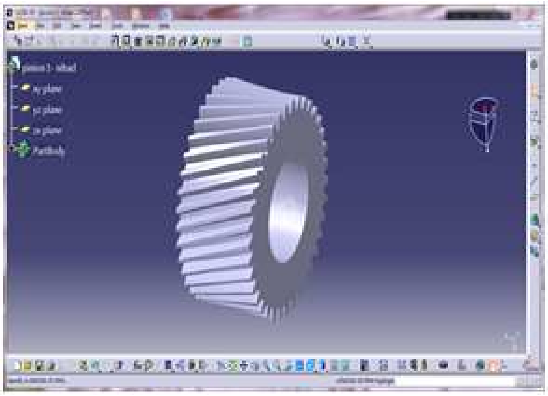

Figure 5: Pinion 1

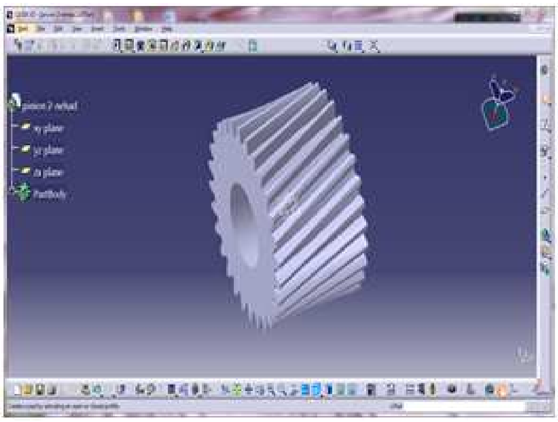

Figure 6: Pinion 2

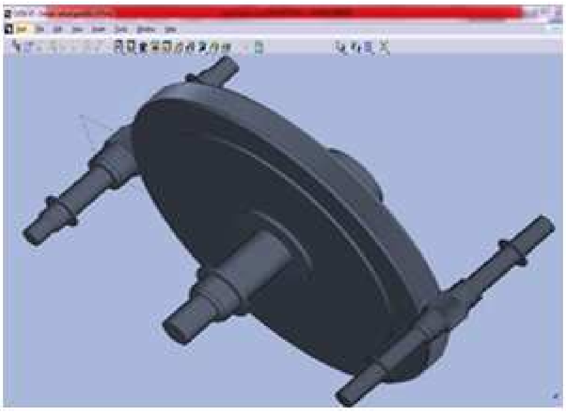

Figure 7: Assembly High Speed Gear

FINITE ELEMENT STRESS ANALYSIS RESULTS

Three different models were tested with single tooth. Model 1 contains full pinion and full gear with one tooth. Model 2 contains full pinion and large part of gear with one tooth. Model 3 contains one tooth of pinion and gear. A small portion of rim is also included into the volume of pinion and gear. Three models of a helical gear drive are shown in the figures. The volume of pinion is divided into three sub-volumes using auxiliary intermediate surfaces. The volume of gear is divided into six sub-volumes using auxiliary intermediate surfaces. Each volume of pinion and gear is discretized by finite elements. The numbers of elements at the possible point of contact and at the root fillet are higher than the other regions. A torque, T = 153.9 N-m is applied to the rotational axis of pinion.

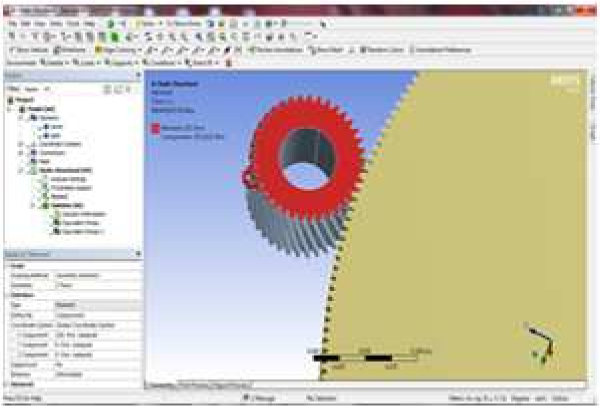

Figure 8: Applied Torque

Table 1

Models	Number of Elements	Number of Nodes	Contact Stress in Pinion(Mpa)	Contact Stress in Gear (Mpa)
Model 1	33,400	37,507	447.601	315.750
Model 2	11,600	13,945	449.553	318.218
Model 3	5,480	6,981	446.881	330.775

From above results, it is visible that there is no variation in the contact stress and hence, it is economical to use one tooth of pinion and gear for contact analysis

Contact Stress

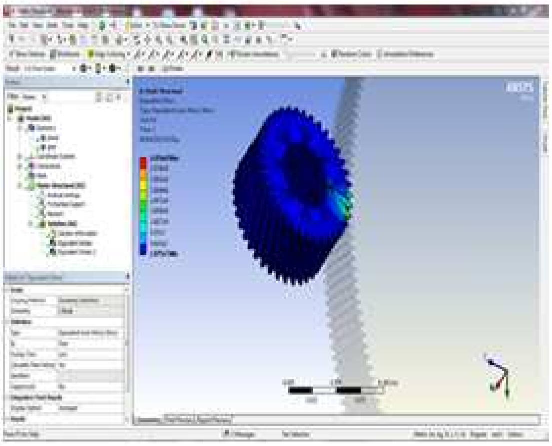

Figure 9: Stresses Contact in Pinion 2

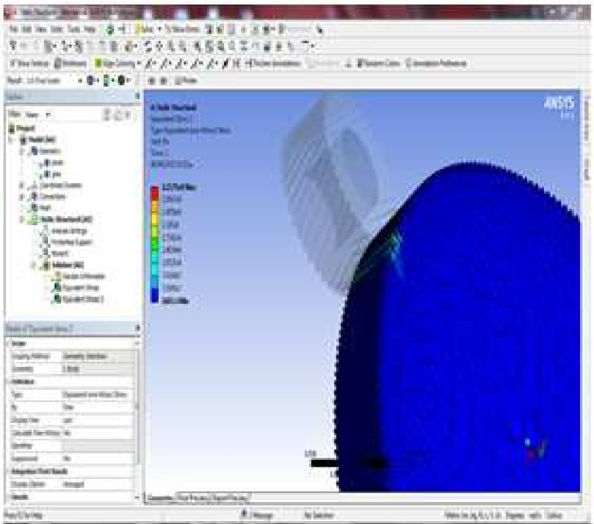

Figure 10: Stresses Contact in Gear

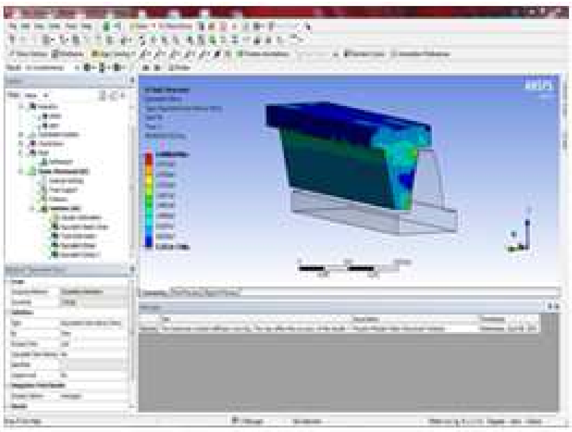

Figure 11: Stresses Contact in Tooth Pinion

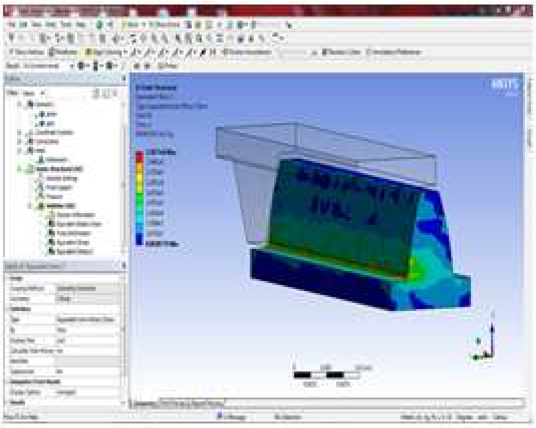

Figure 12: Stresses Contact in Tooth Gear

Table 2

No	Contact Stress in AGMA	Contact Stress in ANSYS	Differences
1	459.128	446.88	2.086 %

To study the contact stress at different location of tooth flank, the pinion is rotated by some angle to get position of contact with gear. The different positions of the point of contact are -3°, 0°, 3°, and 7°. The edge contact in (7 and -3) can take place twice during the engagement of a pair of teeth. The first time the tip of the pinion tooth impacts the root of the gear tooth. Then the edge contact takes place at the end of two teeth

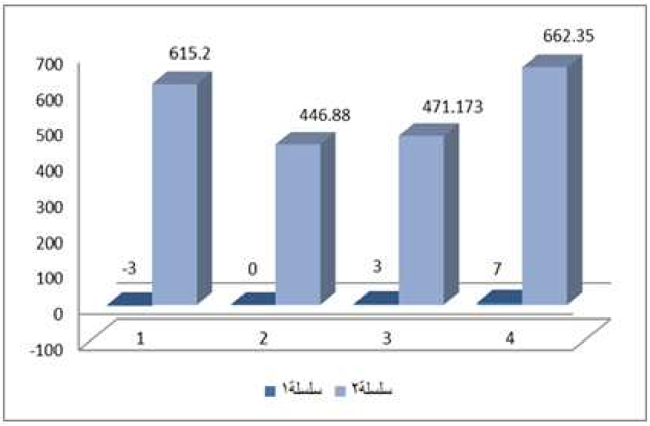

Figure 13

Bending Stresses

Generally, the bending stresses in the fillet of the two contacting tooth side are considered tensile stresses and those in the fillets of the opposite, unloaded tooth side, are considered compressive stresses. The fillet stresses are determined at three pinion's rotational angles 0, 7and -3

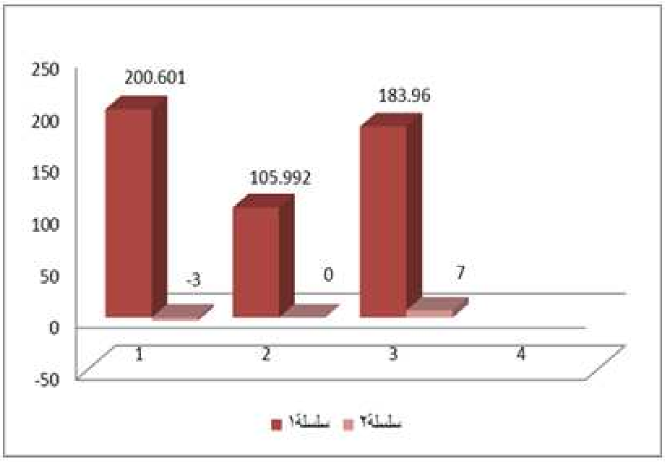

Figure 14

Table 3

no	Bending Stress in AGMA	Bending Stress in ANSYS	Differences
1	93.553	105.992	10.7%

Center Distance Variation

The center distance variations may have occurred in the process of the assemblyThe dislocation bearing contact will result in increase of the bending stress and contact stress.

The contact stress after the center distance variation of 1mm in pinion is 463.011MPa

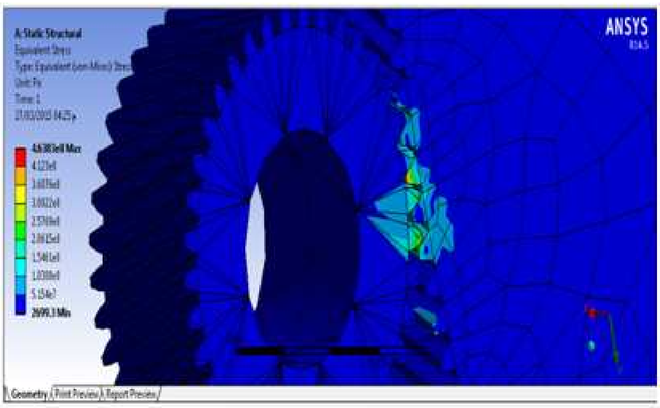

Figure 15

CONCLUSIONS

In this study, finite element stress was performed to investigate the contact stresses and the bending stresses of an helical gear set as mentioned above. The geometrical model of a helical gear was generated using parametric equation of involute curve.

Commercial FEA software, ANSYS workbench 14.5.7

- The contact stresses of an helical gear calculatedAGMA standardis close to the contact stress obtained from ANSYS, And contact stresses are below the allowable contact stresses there for the gear is safe in contact stresses
- The bending stresses are below the allowable bending stresses. Hence the gear is safe in bendingThe bending stresses of an helical gear calculated AGMA standard is close to the bending stress obtained
- The effect of center distance variation and axial misalignment were studied. The center distance variation creates dislocation of bearing contact which results in increase of bending stress and contact stress The axial misalignment will create dislocation of bearing contact
- Shafts are designed, in ch-8 by taking all axial, radial and tangential forces including the weight of the gear into consideration. A factor of safety of 2 taken for calculations. The deflection value of the shaft are under the limit so, design is safe under high speed rotation

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